NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

REPORT No. 341

THE DESIGN AND DEVELOPMENT OF AN AUTOMATIC INJECTION VALVE WITH AN ANNULAR ORIFICE OF VARYING AREA

By WILLIAM F. JOACHIM, CHESTER W. HICKS
and HAMPTON H. FOSTER

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AERONAUTICAL SYMBOLS

1. FUNDAMENTAL AND DERIVED UNITS

	Symbol	Metric		English	
		Unit	Symbol	Unit	Symbol
Length	l	meter	m	foot (or mile)	ft. (or mi.)
Time	t	second	s	second (or hour)	sec. (or hr.)
Force	F	weight of one kilogram	kg	weight of one pound	lb.
Power	P	kg/m/s	----------	horsepower	hp
Speed	----------	km/hr	k. p. h.	mi./hr.	m. p. h.
		m/s	m. p. s.	ft./sec.	f. p. s.

2. GENERAL SYMBOLS, ETC.

W, Weight, $=mg$

g, Standard acceleration of gravity $=9.80665$ m/s$^2=32.1740$ ft./sec.^{2}

m, Mass, $=\frac{W}{g}$

ρ, Density (mass per unit volume).

Standard density of dry air, 0.12497 (kg-m^{-4} s^2) at 15° C and 760 mm $=0.002378$ (lb.-ft.$^{-4}$ sec.^{2}).

Specific weight of "standard" air, 1.2255 kg/m$^3=0.07651$ lb./ft.^{3}

mk^2, Moment of inertia (indicate axis of the radius of gyration, k, by proper subscript).

S, Area.

S_w, Wing area, etc.

G, Gap.

b, Span.

c, Chord length.

b/c, Aspect ratio.

f, Distance from C. G. to elevator hinge.

μ, Coefficient of viscosity.

3. AERODYNAMICAL SYMBOLS

V, True air speed.

q, Dynamic (or impact) pressure $=\frac{1}{2}\rho V^2$

L, Lift, absolute coefficient $C_L=\frac{L}{qS}$

D, Drag, absolute coefficient $C_D=\frac{D}{qS}$

C, Cross-wind force, absolute coefficient $C_C=\frac{C}{qS}$

R, Resultant force. (Note that these coefficients are twice as large as the old coefficients L_C, D_C.)

i_w, Angle of setting of wings (relative to thrust line).

i_t, Angle of stabilizer setting with reference to thrust line.

γ, Dihedral angle.

$\rho\frac{Vl}{\mu}$, Reynolds Number, where l is a linear dimension.

e. g., for a model airfoil 3 in. chord, 100 mi./hr. normal pressure, 0° C: 255,000 and at 15° C., 230,000;

or for a model of 10 cm chord 40 m/s, corresponding numbers are 299,000 and 270,000.

C_p, Center of pressure coefficient (ratio of distance of C. P. from leading edge to chord length).

β, Angle of stabilizer setting with reference to lower wing, $=(i_t-i_w)$.

α, Angle of attack.

ϵ, Angle of downwash.

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Langley Memorial Aeronautical Laboratory

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

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THE DESIGN AND DEVELOPMENT OF AN AUTOMATIC INJECTION VALVE WITH AN ANNULAR ORIFICE OF VARYING AREA

By William F. Joachim, Chester W. Hicks, and Hampton H. Foster

SUMMARY

The injection valve described in this report was designed and developed at the Langley Memorial Aeronautical Laboratory of the National Advisory Committee for Aeronautics in connection with a general research on aircraft oil engines. The purpose of this investigation was to provide an automatic injection valve of simple construction which would produce a finely atomized oil spray of broad cone angle and would fulfill the requirements of fuel injection in aircraft oil engines. The injection valve designed has only six parts—i. e., two concentric nozzle tubes flared at one end, two body parts, and two nuts. The nozzle tubes are provided with seats at the flared ends to form an annular orifice which automatically varies in area with the injection pressure. Adjustment of the nuts determines the valve-opening pressure. The fuel passage to the orifice is provided by the clearance space between the nozzle tubes. When sufficient oil pressure is developed by the fuel pump, the flared ends of the nozzle tubes move apart slightly, and the oil passes through the annular orifice, producing a broad conical spray. The nozzle tubes are so constructed as to cause the cylinder gases to heat them approximately 500° F., which preheats the oil and tends to reduce the ignition lag.

The results of tests made with the N. A. C. A. Spray Photography Equipment on this injection valve indicate the effect of several factors on spray penetration. For a duration of injection of 0.003 second, and a valve-opening pressure of 2,500 pounds per square inch, a change of injection pressure from 6,000 to 10,000 pounds per square inch increased the penetration 25 per cent. For a constant speed and fuel quantity per cycle a change of valve-opening pressure from 2,000 to 5,000 pounds per square inch, which caused a corresponding change in maximum injection pressure from 6,700 to 10,500 pounds per square inch, increased the penetration 5 per cent. A change of spray-chamber air density corresponding to a change of compression ratio of from 11.2 to 15.3 decreased the spray penetration 8 per cent. Curves are presented showing these effects together with the effect of engine-operating temperature on the valve-opening pressure.

Analysis and engine tests indicate that the fuel spray from this type of injection valve has characteristics which reduce the time lag of autoignition and promote efficient combustion in high-speed oil engines.

INTRODUCTION

The design and development of fuel-injection valves to meet the fundamental injection and combustion requirements of high-speed oil engines is a part of the research program of the National Advisory Committee for Aeronautics for the development of the aircraft oil engine. The oil engine has proven its suitability as an efficient power plant when operated at slow speed, but, in order to utilize its inherent advantages at aircraft engine speeds, it requires an injection system that is capable of controlling the injection timing, injection rate, atomization, and the distribution of the fuel injected into the combustion chamber. In addition to these requirements, the time interval between the start of the injection of fuel into the engine cylinder and the start of burning must be of the order of 0.001 second. During this time the oil particles to be ignited must be heated, partially vaporized and the vapor heated to the ignition temperature. If the oil injected into the combustion chamber of an engine cylinder has been well prepared and distributed, it will burn at practically constant volume when ignition occurs. If an entire fuel charge has been injected before ignition occurs, a large amount of constant-volume combustion and dangerously high cylinder pressures result. Thus, the shorter the ignition lag, the more readily the maximum cylinder pressures may be reduced by properly controlling the timing and rate of injection without sacrificing power and economy. Reduction of the ignition lag may be obtained by the use of high compression ratios, correct fuel and air distribution, fine atomization, and the preheating of the fuel. The last three may be accomplished by means of the injection valve.

"Oil sprays from injection valves which depend entirely upon forcing the oil fuel through small round holes to break it up, are finely atomized near the surface, especially at the spray tip, but have a core which is but little atomized." (Reference 1.) With

the above requirements in mind, an automatic fuel-injection valve (fig. 1) was designed and constructed to give a comparatively straight flow path through the valve to an annular orifice. The valve as designed and tested has no moving parts other than

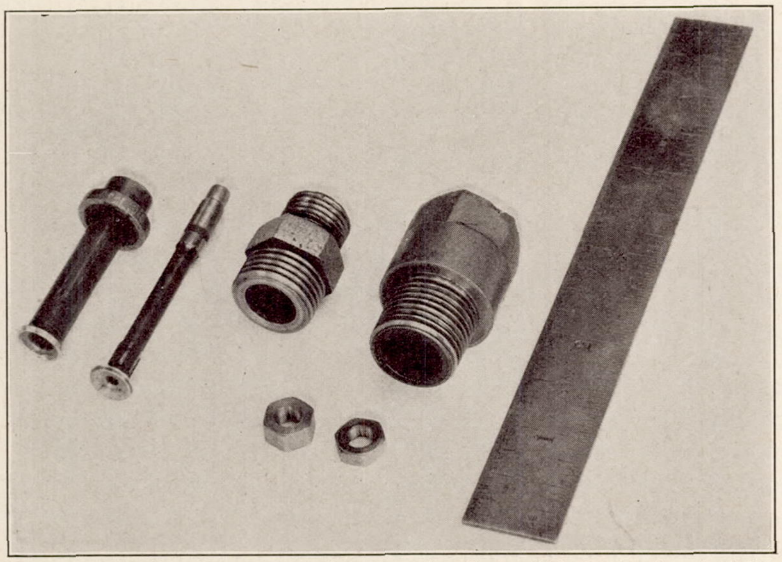


FIGURE 1.—Annular orifice automatic injection valve disassembled showing size, simplicity, and small number of parts

two concentric nozzle tubes which deflect axially to produce an annular orifice.

Tests were made with the N. A. C. A. single-cylinder test engine (Reference 2) to determine the engine performance with this injection valve and also the effect of heat and carbon formation on the nozzle tubes (Reference 3). Some of these data are presented and discussed together with the development

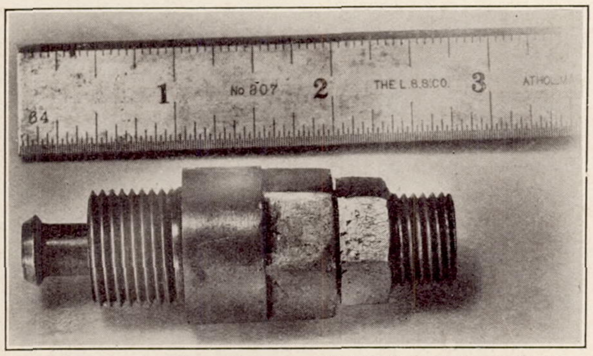


FIGURE 2.—Annular orifice automatic injection valve assembled

of the orifice, seat width, and contour. Tests were made with the N. A. C. A. Spray Photography Equipment (Reference 4) to determine the spray characteristics as affected by changes in injection pressure, valve-opening pressures, and spray-chamber air densities. The results of these tests are also included. An analysis of the design and construction of the injection valve are presented.

DESIGN

In the design of this automatic injection valve, consideration was given to the following mechanical features: Smallest possible number of parts; simple construction; ease of adjustment; automatic, positive, and consistent operation; and an annular orifice of varying area. Special attention was given to the selection of the proper material and the heat treatment, workmanship, clearances, and finish of the valve parts.

Provision was made to give a comparatively straight uninterrupted course through the valve in order to utilize the hydraulic energy in the oil and obtain a finely atomized conical spray, having a high velocity and sufficient penetration to distribute the fuel throughout the combustion chamber under engine-operating conditions.

A means of preheating the oil just before its injection into the cylinder was provided by exposing the outside surface of the outer tube and the inside surface of the inner tube to the hot cylinder gases. The ratio of heating surface exposed to the cylinder gases and that exposed to the oil is approximately 0.96.

Figures 1 and 2 show the injection-valve parts and the assembled injection valve. Figure 3 shows a longitudinal sectional drawing of the valve. Fuel oil is delivered to the upper end of the inner nozzle tube bore as shown by the dotted lines in Figure 3; it then flows through six equally spaced holes to the outside surface to fill the annular space formed between the inner and outer nozzle tubes. As the injection pressure builds up in the fuel line and in this annular space, the two concentric nozzle tubes and the orifice lips formed by their respective flared lower ends are deflected in opposite directions along their common axis, thereby opening the orifice. This axial deflection for full-load fuel quantity is between 0.0005 and 0.0006 inch.

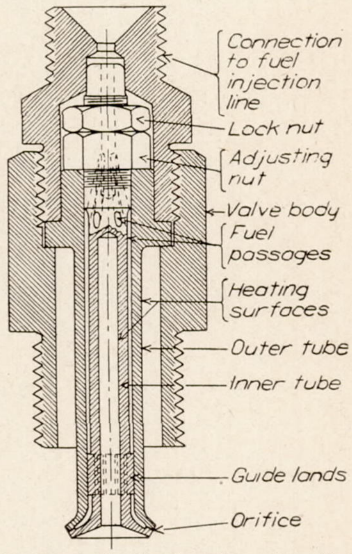


FIGURE 3.—Section through annular orifice automatic injection valve

TESTS

Fuel:

The fuel used in these tests was a commercial grade of Diesel engine fuel oil having a Saybolt viscosity of 41 seconds and a specific gravity of 0.85 at 80° F.

Bench Tests:

Bench tests with the N. A. C. A. fuel-injection test equipment (Reference 5) were made to determine

the injection lag, the duration and cut-off of injection, and the effects of injection-tube length, on the valve-opening pressure. These tests were made by obtaining spray records on blotting paper fastened to the rotating flywheel of the testing equipment. Tests were also conducted to determine the effect of engine-operating temperatures on the valve-opening pressure.

The N. A. C. A. Spray Photography Equipment was used to determine the effect on spray penetration of injection pressures, valve-opening pressures, and spray-chamber air pressures. Figure 4 shows three series of pictures of the spray records obtained during this investigation.

Variable Injection Pressure:

The effect of variable injection pressure on the penetration of the sprays in air under pressure is shown in Figures 5, 6, and 7. Figure 5 shows that for an increase of injection pressure from 6,000 to 8,000 pounds per square inch and a duration of injection of 0.003 second, there is a 19 per cent increase in penetration, and that for an increase from 6,000 to 10,000 pounds per square inch there is a 25 per cent increase in penetration, the spray-chamber air pressure being 150 pounds per square inch gauge. For 180 pounds per square inch spray-chamber air pressure, Figure 6, the increase in penetration is 19 and 28 per cent for an increase in injection pressure from 6,000 to 8,000 and from 6,000 to 10,000 pounds per square inch, respectively. For 210 pounds per square inch spray-chamber air pressure, Figure 7, the increase in penetration is 16 per cent and 26 per cent for an increase in injection pressure from 6,000 to 8,000 and from 6,000 to 10,000 pounds per square inch, respectively.

Effect of Engine-Operating Temperature on the Valve-Opening Pressure:

The upper curve of Figure 8 shows the relation of valve-opening pressure to maximum recorded injection pressure. The lower curve shows the increase in cold-valve-opening pressures caused by the hot cylinder gases during engine operation. The upper curve is plotted from bench test data. Maximum injection pressures were recorded during engine operation for known values of cold-valve-opening pressures and the same constant rates of injection as obtained in the bench tests. The valve-opening pressures corresponding to these maximum injection pressures during engine operation are obtained from the upper curve. These values of opening pressure for the valve heated during engine operation were plotted against the values of opening pressure of the valve when unheated to obtain the lower curve. It may be seen that the ratio of the hot valve-opening pressures to the cold-valve-opening pressures decreases with an increase in cold-valve-opening pressure. With the unheated valve at 300 pounds per square inch valve-opening pressure, the corresponding heated-valve-opening pressure is 1,275 pounds per square inch, or an increase of 325 per cent. For 4,000 pounds per square inch unheated-valve-opening pressure, the corresponding heated-valve-opening pressure is 5,000 pounds per square inch or a 25 per cent increase.

Variable Valve-Opening Pressure:

The effect of varying the valve-opening pressure and the corresponding increase in injection pressure on spray penetration for a constant speed and fuel quantity per cycle are shown in Figures 9 and 10. At a spray-chamber air pressure of 150 pounds per square inch gauge and a duration of injection of 0.003 second, an increase of valve-opening pressure from 2,000 to 3,500 pounds per square inch, which causes a corresponding change in full-load maximum injection pressure on the engine from 6,700 to 9,000 pounds per square inch, resulted in an increase in penetration of only 1.3 per cent. Similarly, an increase in valve-opening pressure from 2,000 to 5,000 pounds per square inch, which causes a corresponding change in maximum injection pressure from 6,700 to 10,600 pounds per square inch, increases the spray penetration only 3 per cent. For a spray-chamber air pressure of 210 pounds per square inch and a duration of injection of 0.003 second, an increase in valve-opening pressure from 2,000 to 3,500 pounds per square inch gave an increased penetration of only 1.4 per cent. An increase from 2,000 to 5,000 pounds per square inch gave 5.1 per cent increase in penetration. The changes in the injection pressure were the same as those at 150 pounds per square inch chamber pressure. The effect of errors in adjusting the valve-opening pressure or in changes in the valve during engine operation has, therefore, a negligible effect on the spray penetration.

Spray Chamber Air Pressure:

The density of the gas into which the spray is injected affects its penetration. (Reference 6.) A survey of Figures 5, 6, and 7 shows a decrease in spray penetration as the chamber pressure is increased. The following are average spray-penetration values for 0.003 second duration of injection, a valve-opening pressure of 2,500 pounds per square inch, and injection pressure of 8,000 pounds per square inch: 2.62-inch penetration for a chamber air pressure of 150 pounds per square inch and 2.38-inch penetration for a chamber air pressure of 180 pounds per square inch or a decrease of 9.16 per cent; and 2.20-inch penetration for

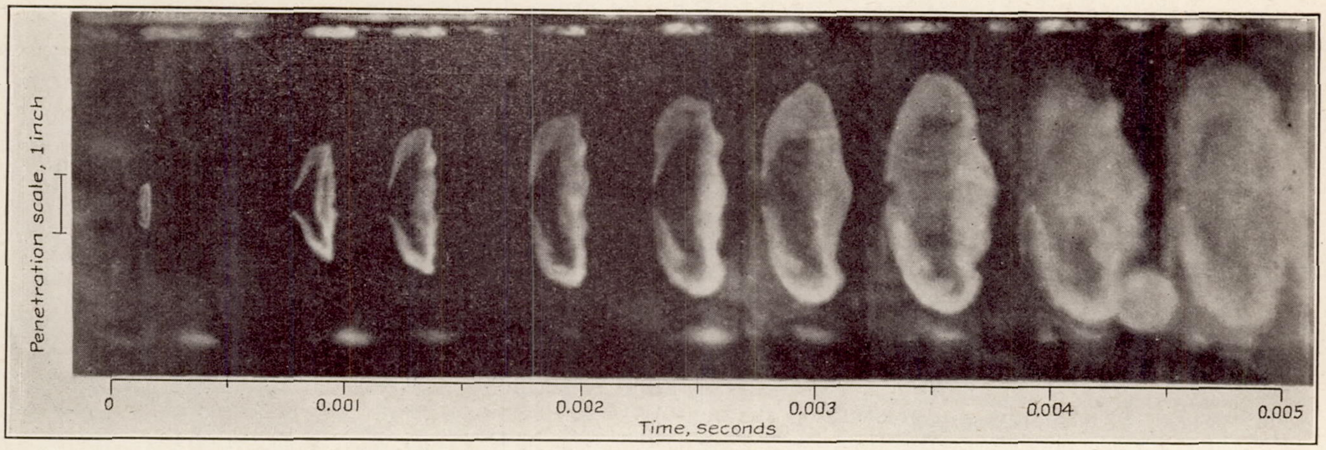


FIGURE 4A.—Injection pressure, 6,000 lb./sq. in. gage. Chamber pressure, 210 lb./sq. in. gage

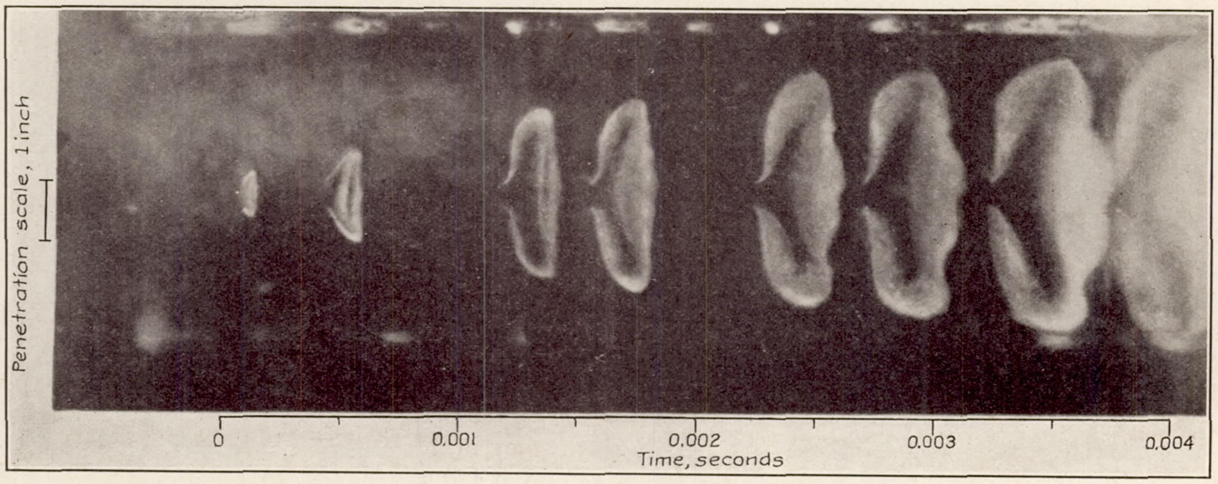


FIGURE 4B.—Injection pressure, 8,000 lb./sq. in. gage. Chamber pressure, 210 lb./sq. in. gage

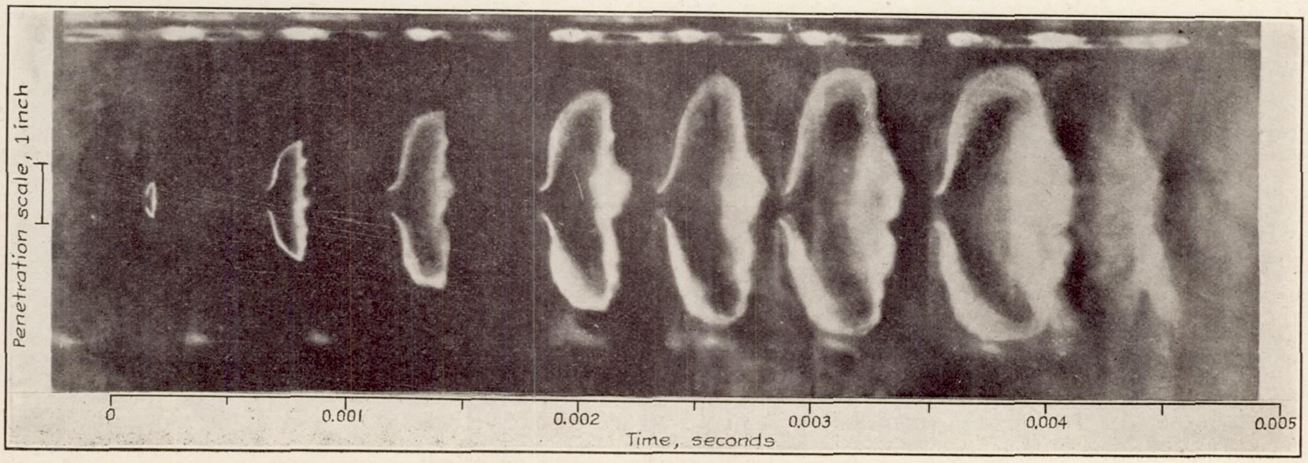


FIGURE 4C.—Injection pressure, 10,000 lb./sq. in. gage. Chamber pressure, 210 lb./sq. in. gage
Spray photographs from annular orifice automatic injection valve

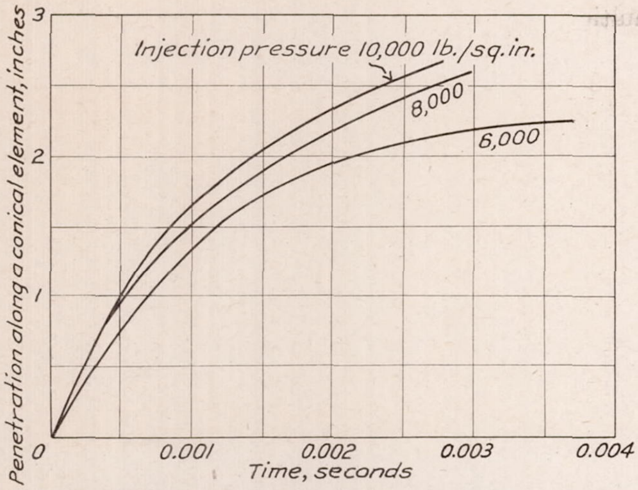


FIGURE 5.—Effect of injection pressure on spray penetration from annular orifice automatic injection valve. Chamber pressure, 150 lb./sq. in. gage. Valve opening pressure, 2,500 lb./sq. in. gage

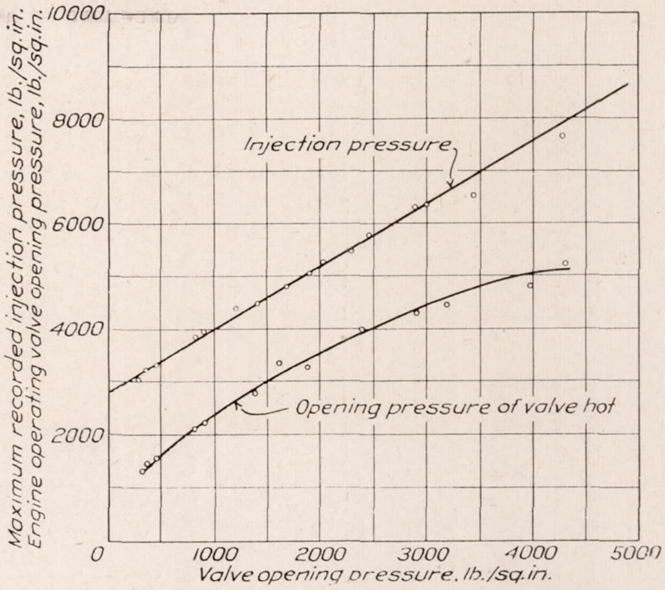


FIGURE 8.—Effect of engine operating temperature on valve opening pressure

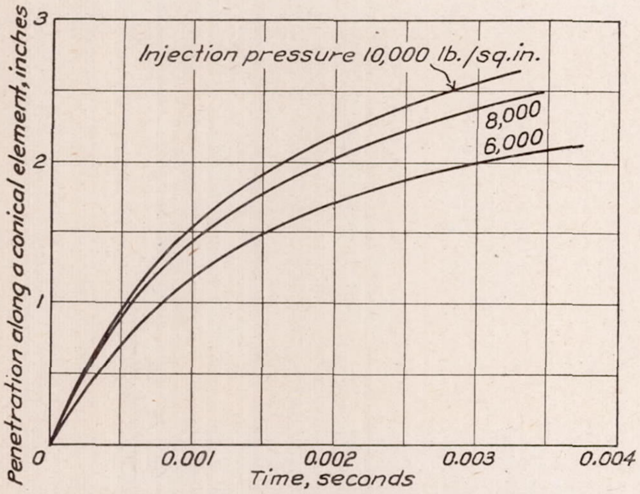


FIGURE 6.—Effect of injection pressure on spray penetration from annular orifice automatic injection valve. Chamber pressure, 180 lb./sq. in. gage. Valve opening pressure, 2,500 lb./sq. in. gage

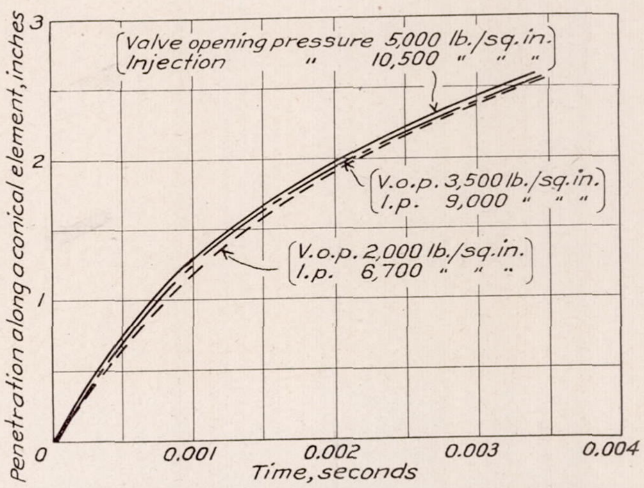


FIGURE 9.—Effect of valve opening pressure on spray penetration from annular orifice automatic injection valve. Chamber pressure, 150 lb. per sq. in. (gage)

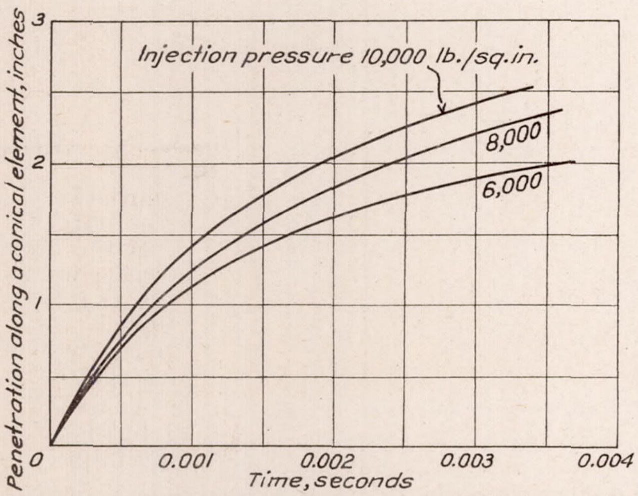


FIGURE 7.—Effect of injection pressure on spray penetration from annular orifice automatic injection valve. Chamber pressure 210 lb./sq. in. gage. Valve opening pressure 2,500 lb./sq. in.

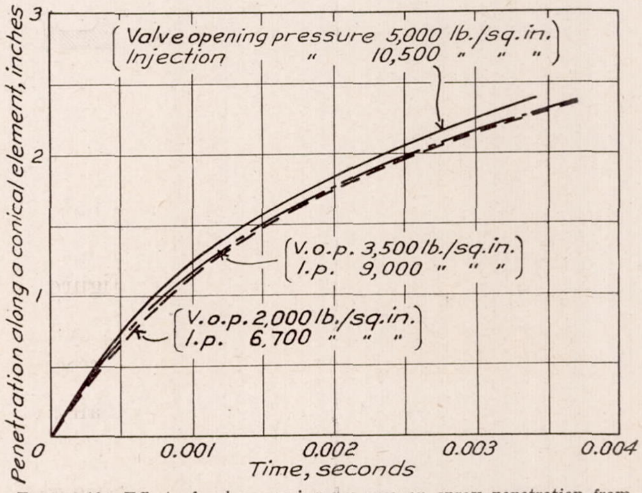


FIGURE 10.—Effect of valve opening pressure on spray penetration from annular orifice automatic injection valve. Chamber pressure 210 lb./sq. in. (gage)

a chamber air pressure of 210 pounds per square inch or a total decrease of 16 per cent from the penetration for a chamber air pressure of 150 pounds per square inch. The density of the air in the spray chamber for pressures of 150 pounds per square inch and 210 pounds per square inch correspond to densities for compression ratios of 11.2 and 15.3, respectively.

Engine Tests:

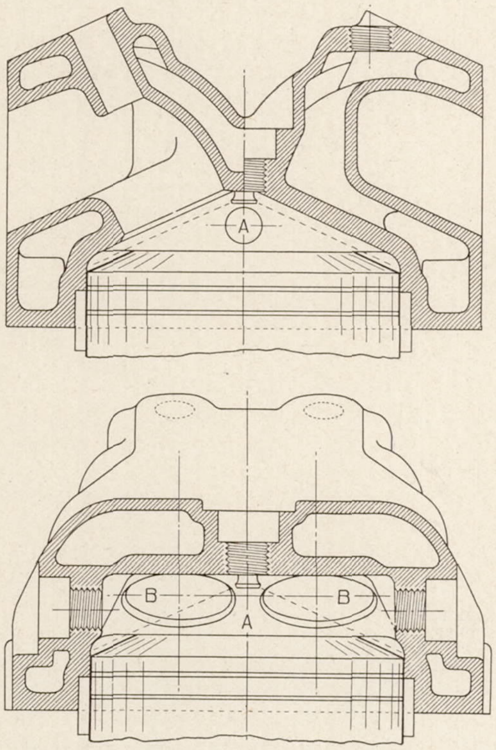
Engine performance tests were made with this injection valve for variable fuel quantity, speed, injection-advance angle, and valve-opening pressure. The results of these investigations will be published in a report to follow. (Reference 3.) Figure 11 shows the typical engine performance obtained and the combustion chamber used in the engine tests with the injection valve described in this report. Right half of Figure 11a is taken from the N. A. C. A. Technical Report No. 282. (Reference 7.) The pent-roof type of cylinder head, used originally for carburetor engine research, was not shaped to fit the spray and the distribution of the fuel was poor, because of the inaccessible air pockets indicated on the figure. The full-load mechanical efficiency of the 5-inch bore by 7-inch stroke single-cylinder test engine is only 75 per cent, because of the large number and size of the auxiliaries driven by the engine. The test data, therefore, have been calculated and plotted for an assumed mechanical efficiency at full load of 85 per cent which is readily attained in multicylinder engines. A multicylinder engine performance of 102 pounds per square inch b. m. e. p. has been computed from data recorded for full-load fuel quantity at 1,750 revolutions per minute. The results of the engine tests indicate that in spite of the type of combustion chamber used, the injection valve gave good performance with low maximum cylinder pressures as indicated by the disk type maximum cylinder-pressure indicator. (Reference 8.)

Mechanical Features:

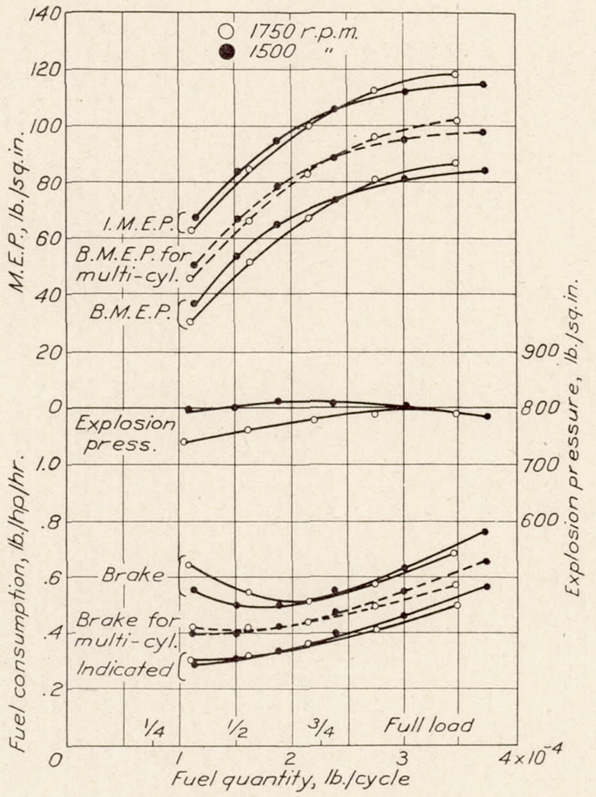
The small movement of the orifice lips of this type of injection valve, a maximum of approximately 0.00055 inch at full load, results in small impact stresses and minimum wear of the moving parts. Stelliting the guide lands resulted in longer life of the nozzle tubes and tended to maintain their concentricity. These features resulted in a uniform spray at all loads without the necessity of frequently grinding or lapping the parts or for replacements. Recent tests of annular orifice fuel-injection valves have shown that the use of disk springs for loading the nozzle tubes would permit of obtaining the desired fuel-injection rates with a smaller range of injection pressures.

Orifice Contour and Seat Development:

Figure 12 shows the several stages of development of the orifice seat and contour. "A" shows the injection valve orifice as first tested. For approximately 10 minutes of operation at 1,500 revolutions per



Combustion chamber of universal test engine showing the air volumes A and B not reached directly by conical spray of the injection valve. Dotted lines show the cone angle of the spray



Fuel-injection performance, universal test engine

FIGURE 11

minute a b. m. e. p. of approximately 118 pounds per square inch was noted but not recorded, due to a steady decrease in performance caused by the flattening out of the orifice lips and the formation of carbon, which caused the change in the spray angle.

"B" shows the first change. The edges of the tubes were machined perpendicular to the lower conical surface of the inner tube, still maintaining line contact at the orifice. These edges could not be properly maintained in engine tests, but the spray angle was more consistent.

"C" shows the seat of the outer tube rounded to give line contact. The seats and guide lands were stellited to prevent undue wear. This decrease in seat width resulted in the indentation of the lower lip with

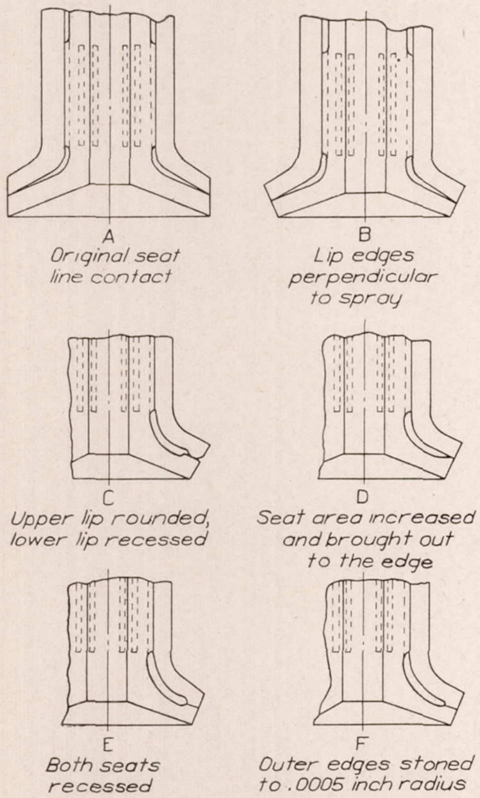


FIGURE 12.—Valve seat development

subsequent spray deflection caused by the hammering action of the lips during seating and in carbon formation at the orifice edge.

"D" shows the seats brought out to the edge, having a width of 0.018 inch. The spray cone angle was slightly uneven and variable because of the formation of a wire edge approximately 0.001 inch or less in width which overlapped the other seat.

"E" shows the tube lips recessed inside to the minimum width requisite for the allowable bearing stress. This shape did not entirely eliminate the formation of the wire edge.

"F" shows the last change made, the outer edges of the lips having been stoned to a 0.0005-inch radius, which resulted in maintaining a constant spray angle.

Spray Formation at Part Loads with Fixed and Variable Orifice Area:

When operating at part loads with a spring-loaded, automatic, fuel-injection valve having a fixed area orifice, the valve stem is raised only a small amount of the total lift while admitting oil to the nozzle chamber. Thus, a longer time will be required to fill the nozzle chamber at part load than will be required at full load, because of throttling through the valve stem and seat. This results in a loss in fuel-spray velocity, penetration, and distribution. The variable area orifice with streamlined flow and practically undiminished hydraulic pressure carried to the inner edge of the orifice gives a high spray velocity, good penetration and distribution at part-load conditions as well as at full load. The thickness of the spray sheet produced by this type of injection valve is extremely fine, which insures that a large surface of fuel spray is presented to the highly heated air in the combustion chamber.

Thermal Preparation of Fuel Spray for Ignition:

Since early autoignition is required for engines to operate efficiently at high speeds and with low maximum cylinder pressures, the fuel must be prepared before and during injection so that it will burn with a small ignition lag. The injection valve described in this report affords a means for preheating the fuel before it is injected into the engine cylinder.

Temper colors on the steel indicate that the lower ends of the nozzle tubes reached temperatures of approximately 450° to 500° F. This increased temperature of the nozzle tubes preheated the fuel before injection so as to lower its surface tension and promote more rapid vaporization and earlier ignition in the engine cylinder. (Reference 9.)

CONCLUSIONS

A comparison of this annular orifice, automatic, injection valve with other types of injection valves shows that it is simple in design and construction. It is small in size, for a 5 by 7 inch engine, about the size of a standard spark plug. It has few parts. It requires only one means of adjustment and is automatic, positive, and consistent in operation. A finely atomized, conical, preheated oil spray of high velocity, providing good penetration and distribution, is obtained at all loads with this injection valve.

The spray photography penetration data show that for an injection duration of 0.003 second and a valve-opening pressure of 2,500 pounds per square inch, a change of injection pressure from 6,000 to 10,000 pounds per square inch increased the penetration 25 per cent. A change in valve opening pressure from 2,000 to 5,000 pounds per square inch increased the spray penetration 5 per cent for conditions of injection such as occur in engine operation. With a constant

injection pressure of 8,000 pounds per square inch and a valve opening pressure of 2,500 pounds per square inch a change of spray-chamber air density corresponding to a change in compression ratio of 11.2 to 15.3 decreased the spray penetration 16 per cent.

Analyses and engine tests indicate that this type of injection valve giving a conical-shaped fuel spray has characteristics which reduce the time lag of auto-ignition and promote efficient combustion in high-speed oil engines.

LANGLEY MEMORIAL AERONAUTICAL LABORATORY,
NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS,
LANGLEY, VA., *July 15, 1929.*

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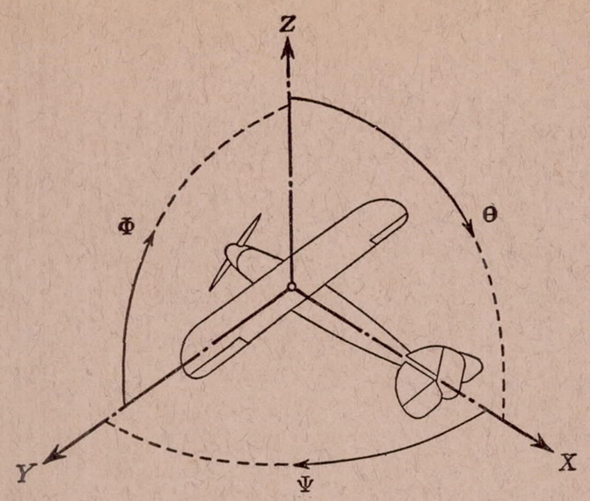

Positive directions of axes and angles (forces and moments) are shown by arrows

Axis		Force (parallel to axis) symbol	Moment about axis			Angle		Velocities	
Designation	Symbol		Designation	Symbol	Positive direction	Designation	Symbol	Linear (component along axis)	Angular
Longitudinal	X	X	rolling	L	$Y \longrightarrow Z$	roll	Φ	u	p
Lateral	Y	Y	pitching	M	$Z \longrightarrow X$	pitch	Θ	v	q
Normal	Z	Z	yawing	N	$X \longrightarrow Y$	yaw	Ψ	w	r

Absolute coefficients of moment

$$C_L=\frac{L}{qbS} \qquad C_M=\frac{M}{qcS} \qquad C_N=\frac{N}{qfS}$$

Angle of set of control surface (relative to neutral position), δ. (Indicate surface by proper subscript.)

4. PROPELLER SYMBOLS

D, Diameter.
p_e, Effective pitch.
p_g, Mean geometric pitch.
p_s, Standard pitch.
p_v, Zero thrust.
p_a, Zero torque.
p/D, Pitch ratio.
V', Inflow velocity.
V_s, Slip stream velocity.

T, Thrust.
Q, Torque.
P, Power.
(If "coefficients" are introduced all units used must be consistent.)
η, Efficiency $=T\,V/P$.
n, Revolutions per sec., r. p. s.
N, Revolutions per minute, r. p. m.
Φ, Effective helix angle $=\tan^{-1}\left(\frac{V}{2\pi rn}\right)$

5. NUMERICAL RELATIONS

1 hp = 76.04 kg/m/s = 550 lb./ft./sec.
1 kg/m/s = 0.01315 hp
1 mi./hr. = 0.44704 m/s
1 m/s = 2.23693 mi./hr.

1 lb. = 0.4535924277 kg
1 kg = 2.2046224 lb.
1 mi. = 1609.35 m = 5280 ft.
1 m = 3.2808333 ft.